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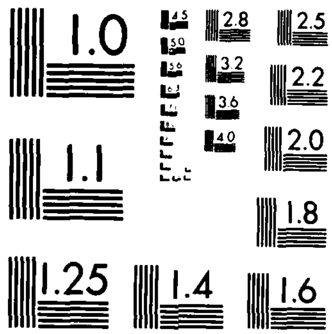

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CONTRACTOR REPORT ARLCD-CR-85002

PRODUCT IMPROVEMENT PROGRAM FOR THE M577 FUZE--VOLUME 6, LOWER COST TRIGGER SHAFTS

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The objective of this project was to reduce the cost of the trigger shafts in M577 fuzes through the use of 2024 aluminum instead of 416 stainless steel. All four shafts in the trigger were tested using aluminum. In the final analysis, two of these, the safety plate shaft and the release lever shaft, were recommended for implementation in aluminum.

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CONTENTS

INTRODUCTION

This report covers a product improvement program aimed at reducing the cost and enhancing the producibility of the M577 fuze. The objective of the work was to reduce the cost of the trigger shafts by substituting aluminum for stainless steel.

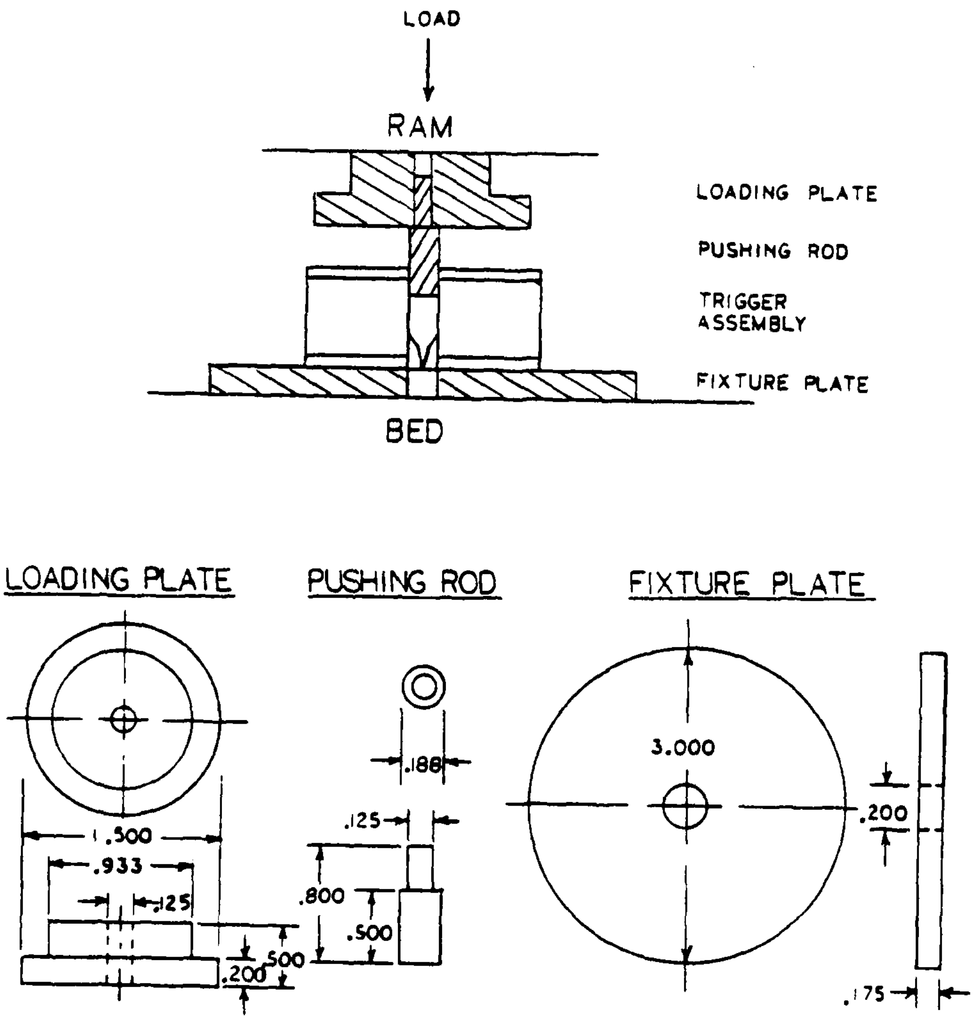
The development of 2024, T4 aluminum trigger shafts is discussed. The material properties and manufacturing methods for the firing arm shaft, release shaft, rotor detent, and safety plate shaft were studied.

DISCUSSION

The material properties and manufacturing processes of various aluminum alloys were studied. Aluminum alloy, 2024, T4, was selected for its high strength, machinability, and availability for the rotor detent, safety plate shaft, and release shaft. It was decided to retain the stainless steel firing arm shaft because of burr removal problems with the aluminum.

The major areas of concern in substituting 2024, T4 aluminum for stainless steel are the strength and rigidity of the aluminum. The trigger shafts are required to withstand the combined loads of 30,000 g acceleration and 30,000 RPM spin and be functional afterwards. Since the safety plate shaft is the most severely stressed of the four trigger shafts, a force analysis of this shaft was performed (See Appendix A). To simplify the calculations, assume the load is applied at the centerline of the safety plate shaft. The shear stress of the safety plate shaft at the bottom pivot was found to be 9,442 psi. The minimum shear strength of 2024, T4 aluminum is 37,000 psi. The calculated safety factor for the proposed aluminum in the worst case is 3.9. In reality, the safety factor would be greater since these calculations were based on assumptions described earlier that increase the calculated stress over the actual stress. These calculations show that substituting 2024, T4 aluminum for stainless steel in the three trigger shafts is feasible. Therefore, static load tests were performed on the 2024, T4 aluminum safety plate shaft, since it is the most severely stressed of the trigger shafts.

Aluminum and stainless steel safety plate shafts were assembled in trigger spacers, and the assembled triggers were loaded through the firing pin using a test fixture to simulate the setback condition of the safety plate assembly as shown in Figure 1. The aluminum safety plate shaft failed by shearing off the bottom pivot of the safety plate shaft at the equivalent load of 114,000 g's, and the stainless steel safety plate shaft failed the same failure at the equivalent load of 226,000 g's. Using 30,000 g's as the design criteria, this indicates an adequate safety factor of 3.8 for the aluminum shaft and 7.5 for the stainless steel shaft. The safety plate shafts with the sheared bottom pivots still performed satisfactorily at 1000 RPM no-fire and 2000 RPM with 600 lb. load must fire tests.



Note: The center hole diameter of Trigger Top Plate was drilled to .200 inch to perform the Static Compression Test.

Figure 1. Static test fixture

All testing, but forty foot drop test, was performed with all four shafts in the trigger. The forty foot drop test was conducted with three aluminum shafts and stainless steel firing arm shaft. The aluminum firing arming shaft was later dropped from consideration because no economical means could be found to remove the chips at the edge of the shaft slot.

The compatibility between aluminum and stainless steel was checked and found to be acceptable. The electrochemical potential relative to galvanic action between aluminum and stainless steel is 0.10 volts. Any galvanic couple having a potential of .25 volts or less is considered acceptable. There are many horological type parts that are not lubricated nor subjected to additive protective finishes. In the M577 fuze there are already stainless steel parts in direct contact with bare aluminum parts in several places; e.g., the gear and pinion, and the pivots and plates in the SSD. This combination has caused no adverse function of the fuze. Therefore, no protective finish is recommended for the aluminum shafts in the trigger.

TESTING

Spin Test

Ten units, with aluminum firing arm shafts, safety plate shafts, release shafts, and rotor detents in the triggers, were built and spin tested at 20,000 to 30,000 RPM. The triggers functioned properly in all units. Unit by unit results are shown in Table 1.

Table 1. Spin test results

Unit	RPM	Result
1	25,000	Fired properly
2	20,000	Fired properly
3	28,000	Fired properly
4	30,000	Fired properly
5	28,000	Fired properly
6	25,000	Fired properly
7	30,000	Fired properly
8	30,000	Fired properly
9	30,000	Fired properly
10	27,000	Fired properly

Air Gun Test

Ten units, containing aluminum trigger shafts, were air gun tested at 29,000 g to 32,000 g. After the test, the fuzes were disassembled and tested satisfactorily in the 1,000 RPM no fire and 2,000 RPM under 600 lb. load must fire tests. No parts in the trigger were damaged from the test. Results are shown in Table 2.

Table 2. Air gun test results

Unit	g Level	Spin Tests Acceptable	Observation
1	31,811	Yes	No damage
2	30,910	Yes	No damage
3	30,552	Yes	No damage
4	30,672	Yes	No damage
5	30,298	Yes	No damage
6	30,342	Yes	No damage
7	31,434	Yes	No damage
8	31,488	Yes	No damage
9	30,788	Yes	No damage
10	29,738	Yes	No damage

Ballistic Test

One hundred twenty fuzes with four aluminum trigger shafts were ballistically tested in September, 1981. The point reliability of the functional tests was 98%. A nonfunction safety test with the fuze set on the shipping setting showed the trigger held the interlock pin in the safe position. A summary of the test results is shown in Table 3. Round by round data were reported by the U.S. Army Yuma Proving Grounds in Firing Report No. 81-PI-0231-L5.

Table 3. Ballistic test results

TPR #2594, Supplement 12

Lot #HAT81H000E063

No. of Units	Gun	Zone	Time Sec.	Environment (°F)	Function	Mean	Std. Dev.
20	155mm, M198	8	105	70	19/20[a]	105.496	.362
20	8 in., M2A2	1	3	70	19/20[b]	3.055	.038
20	155mm, M185	8	3	70	20/20	3.078	.046
20	105mm, M103	7	3	145	20/20	3.132	.042
20	105mm, M103	7	Ship Set	70	0/20	Must not fire - 820 ft.	
20	155mm, M107	1	PD	70	20/20	Must fire - 820 ft.	

a. Dud did not function ground impact and was not recovered.

b. Dud functioned on ground impact.

Forty-Foot Drop Test

Five fuzes with three aluminum trigger shafts were subjected to the Forty-Foot Drop Test per MIL-STD-331, Test 103. All units were examined and found to satisfy the requirements of MIL-F-50983 and be safe to handle and dispose of following testing:

COST & WEIGHT

Cost Comparison

A cost comparison of the stainless steel and aluminum trigger shafts is shown in Table 4. The lower cost, using aluminum, results from lower material cost and faster cycle time. The costs of the trigger shafts are based on a quantity of 300,000 units. The projected cost of the tooling is approximately $1,500. The projected cost savings of $.1119 per fuze for the three trigger shafts does not include general and administrative expenses, tooling and profit.

Table 4. Cost comparison*

	416 Stainless Steel ($)	2024 Aluminum ($)	Savings ($)
Safety Plate Shaft	.0763	.0604	.0159
Release Shaft	.1090	.0723	.0367
Rotor Detent	.1792	.1199	.0593
Total			.1119

* 1982 dollars.

Weight

The weight reduction from substituting three aluminum shafts for stainless steel shafts is .0011 lbs., which is negligible.

CONCLUSIONS AND RECOMMENDATIONS

Fuzes made with 2024, T4 aluminum trigger shafts were subjected to spin, air gun, environmental, and ballistic tests with excellent results. It was recommended that 2024, T4 aluminum be substituted for stainless steel in the rotor detent, safety plate shaft, and release shaft and that the current firing arm shaft be retained. However, problems were experienced in trying to implement the rotor detent. A rejection rate of 20% occurred during implementation in the 2,000 RPM spin test versus a 4% rejection rate during development. After a lengthy investigation, no explanation could be found for this difference. It is, therefore, now recommended that the stainless steel rotor detent be retained.

APPENDIX A

STRESS ANALYSIS

Since the safety plate shaft is the most severely stressed of the four trigger shafts, the stress analysis of this shaft is shown in details:

1. Force Analysis of Safety Plate Shaft

The combination of the force due to 30,000-g acceleration (of insert, spring, firing pin and safety plate assembly) and the force due to 30,000-RPM spin (of the safety plate assembly) will be applied to the Safety Plate Shaft at the instant the projectile is fired.

a. Force due to 30,000-g acceleration of insert, spring and firing pin

Force $= m \cdot a = (W/g)(30{,}000\ g) = 30{,}000\ W$, in lbs.

where m = mass in slugs

W = weight in lbs.

a = linear acceleration in ft./sec.^{2}

g = gravitational acceleration

$= 32.2$ ft./sec.^{2}

therefore,

Force $= 30{,}000\ W$, (See table 5 for the weight.)

$= 30{,}000\ (.0001 + .0005 + .0014)$

$= 60$ lbs.

b. Force due to 30,000-g acceleration of Safety Plate Assembly

Force $= m \cdot a$

$= 30{,}000\ W$

$= 30{,}000\ (.0015)$

$= 45$ lbs.

Table A-1. Weight comparison

Part or Assembly	Current Design Stainless Steel Shaft		Proposed Design Aluminum Shaft	
	Weight, Lbs.	Mass, Slug	Weight, Lbs.	Mass, Slug
Top Plate	0.0328			
Bottom Plate	0.0230			
Spacer with 4 Pins	0.0625			
Insert	0.0001			
Firing Pin Spring	0.0005			
Firing Pin	0.0014			
Firing Arm Ass'y.	0.0048	0.0001491	*(0.0032)	*(0.0000994)
Rotor Detent Ass'y.	0.0010	0.0000311	0.0008	0.0000248
Release Lever Ass'y.	0.0016	0.0000497	0.0010	0.0000311
Safety Plate Ass'y.	0.0018	0.0000560	0.0015	0.0000466
Setback Pin	0.0005			
Setback Spring	0.0001			
Setback Retainer	0.0001			
Two Rivets	0.0026			
TOTAL	0.1328			
Trigger Assembly	0.1321			
Firing Arm	0.0011			
Rotor Detent	0.0006			
Release Lever	0.0007			
Safety Plate	0.0013			

NOTES: 1. The weight difference between Trigger Assembly (0.1321 lbs.) and Total (0.1328 lbs.) indicates the inaccuracy of measurement and/or variation of parts weight.

*2. The aluminum Firing Arm Shaft is not the recommended design.

c. Force due to 30,000-RPM spin of Safety Plate Assembly

Normal Force = $m r \omega^2$, in lbs.

Tangential Force = $m r \alpha$, in lbs.

where m = mass in slug

r = radius in ft.

ω = angular velocity in radians/sec.

α = angular acceleration in radians/sec.^{2}

therefore,

$$\begin{aligned} \text{Normal Force} &= m \cdot r \cdot \omega^2 \\ &= (.0000466)(.02)(3141.6)^2 \\ &= 9.20 \text{ lbs.} \end{aligned}$$

$$\begin{aligned} \text{Tangential Force} &= m \cdot r \cdot \alpha \\ &= (.0000466)(.02)(1545600) \\ &= 1.44 \text{ lbs.} \end{aligned}$$

(See Appendix B for values of ω and α .)

d. Direction of Loading

The center of the Safety Plate Shaft is assigned as "0" position of X-Y axis, and the midpoint of the assembled safety plate thickness is assigned as "0" position of Z axis. (See the Force Analysis Diagrams shown in Figures 2 and 3.)

e. Load in X-Y Direction

There are two forces acting on the Safety Plate Assembly in X-Y direction. One is a force of the safety plate assembly due to 30,000-RPM spin, and the other is a reaction force in X-Y direction due to the force described in "article a."

The reaction force in the X-Y direction may be found as follows:

$R = Pz \cdot \tan \theta$,

where R = reaction force

Pz = force described in "article a."

θ = inclined angle of firing pin

then,

$R = 60 \tan 20^{o}$

$= 21.8$ lbs.

It is assumed that the direction of the tangential and normal force due to spin is the same. To simplify the calculation further, the direction of the reaction force and the normal force is assumed the same, which is the most severe case.

Then the load in the X-Y direction can be added, and the total force is as follows:

Pxy = Normal Force + Tangential Force + Reaction Force

$= 9.2 + 1.44 + 21.8$

$= 32.44$ lbs.

f. Load in Z Direction

When the projectile is fired, the force due to 30,000-g acceleration of insert, spring, firing pin and the safety plate assembly is in the direction of the Z axis.

To simplify the calculations, assume the load is applied at the centerline of the safety plate shaft. This is a conservative assumption.

Then 60 lbs. will be loaded to the safety plate with 0.199 inch distance as shown in Figure 3. (0.199 inch may be obtained from $\sqrt{a^2 + b^2}$ in Figure 2.)

g. Action and Reaction of the Safety Plate Assembly

The total force Pxy (32.44 lbs.) will act on the safety plate assembly in X-Y direction, causing reactions at the top and bottom of the Safety Plate Shaft and at the point where the safety plate contacts the Firing Arm Shaft. (See Figure 2.)

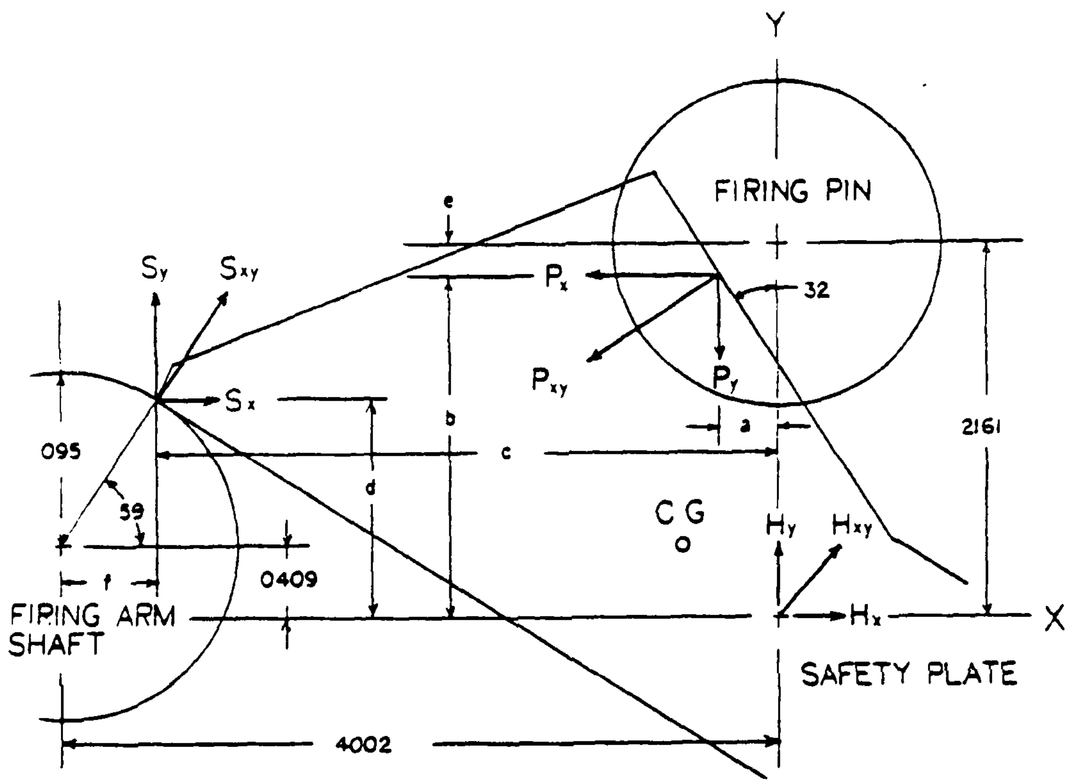


Figure A-1. Force Analysis Diagram (X-Y Direction)

where
a = .031, measured
b = .2161 - e = .2161 - .031 tan 32
= .1967
c = .4002 - f = .4002 - .095 cos 59
= .3513
d = .0409 + .095 sin 59
= .1223

NOTES:

1. 32° and 59° angles and dimension "a" were measured from a 12x layout drawing.

2. C.G. is the approximate center of gravity in Safety Plate Assembly.

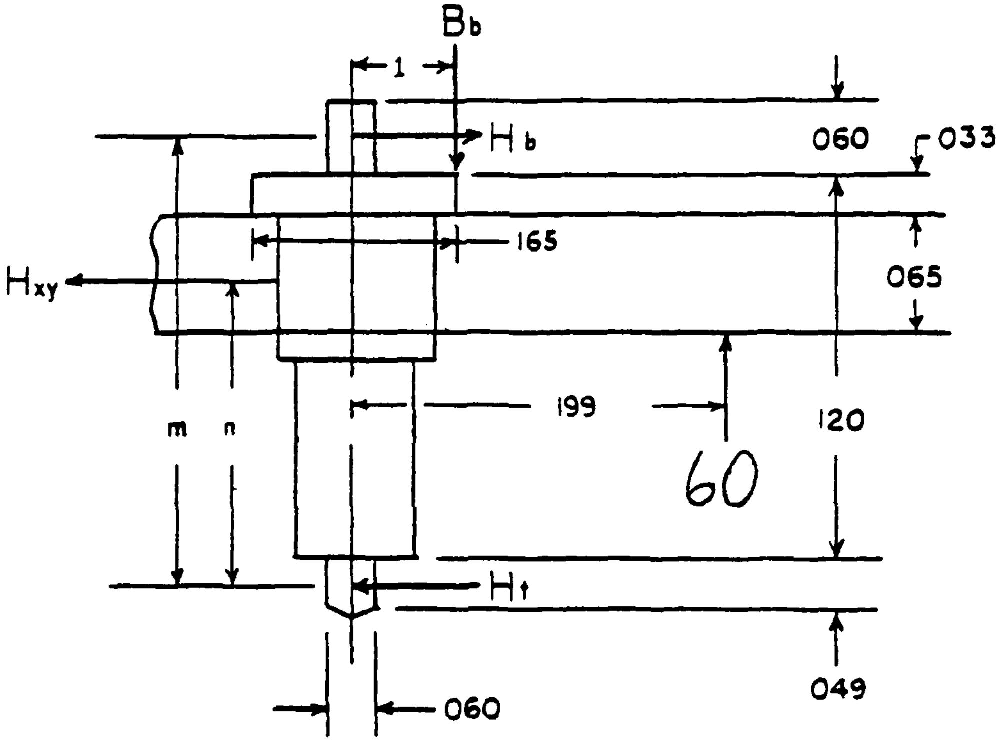


Figure A-2. Force Analysis Diagram (Z Direction)

where $1 = .165/2 = .0825$
$m = .120 + (.060 + .049)/2 = .1745$
$n = m - .033 - (.060 + .065)/2 = .079$

In the Z direction, (60 + 45) lbs. will act on the safety plate causing reactions at the top and bottom of the Safety Plate Shaft and at the shoulder of the Safety Plate Shaft. (See Figure 3.)

2. Stress Analysis of the Safety Plate Shaft

a. Reaction of the Safety Plate Shaft due to total force (Pxy)

From the diagram in Figure 2, the following equations may be obtained.

$$Fx = 0 = -Px + Sx + Hx$$

$$Fy = 0 = -Py + Sy + Hy$$

where $Px = Pxy \cos 32 = 32.44 \cos 32 = 27.5$ lbs.

$Py = Pxy \sin 32 = 32.44 \sin 32 = 17.2$ lbs.

$$M_H = 0 = -Px(b) - Py(a) + Sx(d) + Sy(c)$$

$$= -27.5(.1967) - 17.2(.031) + .1223\ Sx + .3513\ Sy$$

$$= -5.41 - 0.53 + .1223\ Sx + .3513\ Sy$$

From the geometry of Sx and Sy,

$Sy / Sx = \tan 59^{\circ} = 1.6643$ or $Sy = 1.6643\ Sx$

Now the moment equation may be solved as follows:

$$M_H = 0 = -5.41 - 0.53 + .1223\ Sx + .3513\ Sy$$

$$= -5.94 + .1223\ Sx + .3513\ (1.6643\ Sx)$$

$$= -5.94 + .7070\ Sx$$

therefore,

$$Sx = 5.94 / .7070 = 8.4 \text{ lbs.}$$

$$Sy = 1.6643 \, Sx = 1.6643 \, (8.4) = 14.0 \text{ lbs.}$$

$$Hx = Px - Sx = 27.5 - 8.4 = 19.1 \text{ lbs.}$$

$$Hy = Py - Sy = 17.2 - 14.0 = 3.2 \text{ lbs.}$$

b. Shear Load at the top and bottom of shaft pivot

To find the shear load on the trigger shaft pivots, the forces in Z direction should be considered. From the diagram in Figure 3, the following equations may be obtained.

$$Fz = 0 = 60 - Bb$$

$$Fxy = 0 = Hb - Ht - Hxy$$

therefore,

$$Bb = 60$$

$$Hb - Ht = Hxy = \sqrt{Hx^2 + Hy^2} = \sqrt{19.1^2 + 3.2^2} = 19.4$$

and,

$$\begin{aligned} Mt = 0 &= -Hxy(n) + Hb(m) + Bb(l) - (60(.199) \\ &= -(19.4)(.079) + Hb(.1745) + 105(.0825) \\ &\quad - (60)(.199) \\ &= -\,4.81 + .1745\, Hb \end{aligned}$$

therefore,

$Hb = 4.81 / .1745 = 27.57$ lbs.

$Ht = Hb - 19.4 = 78.9 - 19.4 = 59.5$ lbs.

c. Stress of the Safety Plate Shaft Pivot in the Bottom Plate

$\sigma = L / A$, where $\quad$ = Stress in psi

L = Load in lbs.

A = Shear Area in $inch^2$

$A = \pi D^2 / 4 = \pi (.061)^2 / 4 = .00202 \text{ in.}^2$

Therefore, the stress at the bottom pivot is

$\sigma = 27.57 / .00292 = 9{,}442$ psi

APPENDIX B

CALCULATION OF ALPHA AND OMEGA (α and ω)

1. Calculation of ω

The value of ω (angular velocity in radians per second) may be found from 30,000-RPM spin as follows:

$$\omega = 30{,}000 \frac{\text{revolutions}}{\text{minute}} = 30{,}000 \frac{2\pi \text{ radians}}{\text{minute}} = 30{,}000 \frac{2\pi \text{ radians}}{60 \text{ seconds}}$$

$$= (30{,}000)(2\pi/60)(\text{radians/sec}) = 3141.6 \text{ (rad/sec)}$$

2. Calculation of α

The value of α (angular acceleration in radians per second squared) may be found as follows:

$$\alpha = \frac{d\omega}{dt} = \frac{d}{dt}(KV) = K\frac{dV}{dt} = K \cdot a$$

where K is a constant and $K = \dfrac{12(2\pi)}{ND}$, Radians/ft

V is the Muzzle Velocity in ft/sec

N is a constant and N = 1/twist

D is the Land Diameter in inches

a is the Linear Acceleration in ft/sec^2

Now the maximum linear acceleration (a) is 30,000 g, and the maximum K results when the value of N times D is the smallest. From the cannon data in Report No. APG-MT-4503 (Methodology Investigation on Setback and Spin by Heppner), the minimum value of N times D is 47.19 for the 40mm gun.

therefore, $= K \cdot a = [12(2\pi)/ND]\, a = 1.60(a) = 1.60(30{,}000\text{ g})$

$$= 1.60(30{,}000)(32.2) = 1{,}545{,}600 \text{ (rad/sec}^2)$$

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